STUDY OF THE EFFICIENCIES AND THE RELATIVE MERITS OF CONVECTION AND RADIANT HEATING SYSTEMS

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Harry Carter Robinson Junior

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AND RADIANT HEATING SYSTEMS

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PREFACE

MEANING OF SYMBOLS AND ABBREVIATIONS USED

Ave.	Average
BTU	British Thermal Unit
Conv.	Convective
F	Fahrenheit or Degrees Fahrenheit
Hrs.	Hours
h_f	Enthalpy of the condensate leaving the convertor, BTU/LB.
h_g	Enthalpy of the steam entering the convertor, BTU/LB.
Max.	Maximum
Min.	Minimum
Mins.	Minutes
MRT_A	Mean Radiant Temperature as indicated by actual surface readings
MRT_V	Mean Radiant Temperature as indicated by the Vernon thermometer
O.G.	Outside Glass (windows)

Q.	Heat Supplied, BTU
R	Rankine(degrees)
Rad.	Radiant
t_a	Ambient temperature in degrees fahrenheit
t_g	Globe or Vernon temperature as read by the thermometer in the globe - degrees fahrenheit
T_s	Surface temperature in degrees fahrenheit
Temp.	Temperature
X	Less than 16 feet per minute (air velocity)

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STUDY OF THE EFFICIENCIES AND THE RELATIVE MERITS OF CONVECTION AND RADIANT HEATING SYSTEMS

I INTRODUCTION

Much has been written, as projected thought, about installed panel heating systems, but little actual data is available.[1] Panel heating, often referred to as radiant heating, has been installed in many buildings throughout this country and in several foreign countries.[2] However, an easy comparison with another type of heating system is not very readily made because of architectural and structural difficulties.

An ideal comparative study would best be afforded by two identical buildings located in the same proximity, so as to have the same exposure to all weather conditions. One building would have a radiant heating system, while the other would have installed another system for comparison.

A very close example of this is the Burge Apartment

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[1] T. Napier, Radiant Heating (New York: The Industrial Press, 1947), p. 65.

[2] B. F. Raber and F. W. Hutchinson, Panel Heating and Cooling Analysis (New York: Johns Wiley and Sons, Inc. London: Chapman and Hall, Ltd., 1947), p. 15.

Building, 210 North Avenue, N.W., Atlanta, Georgia, where this investigation was conducted. However, it is not precisely an ideal building for a comparative study of two heating systems. Actually, the structure is two entirely separate buildings for all practical heating purposes. The building, which is "H" shaped, is divided into two similar sections by a vertical wall from the basement to the roof. One section has installed a panel heating system, whereas in the other there is a convection heating system. This building is eight story in height, with eight apartments on each floor, making a total of sixty-four apartments. That part of the building containing the radiant panels is larger than the other by 2,478 square feet of heated floor space. The overall heated floor space is 45,934 square feet, showing a variance of 5.4% of floor space. As an offset to this extra heating load required by the panel heated apartments, they are afforded a larger southern exposure, presenting the benefit of an increase of solar radiation over the northern portion of the building which contains convection heating units in each room.

That portion of the building heated by the panel system has no visible heating units in any of the rooms, for hot water pipes are buried in each floor of every room. The floors are all of 5-inch concrete, with plaster on the under side, while they are covered with asphalt tile above. The hot water pipes, 1-inch nominal diameter, are located at a depth of $2\frac{1}{2}$ inches in the concrete.

The convection system is quite different, as a radiator type unit, a finned coil, installed in each room, is equipped with a metal case to increase the natural draft of heated air, thereby setting up convective currents in the room. This, too, is a hot water system and in no case is steam brought into any of the rooms.

The hot water for both sections is heated in the basement by two separate 5,000 gallon per hour convertors, which use low pressure steam as the heat source. Each convertor has its individual expansion tank and individual condensate meter in order that the steam requirements of both heating units can be computed independently.

This investigation of the two types of heating systems is of paramount interest to the heating engineer from many standpoints, especially economy. In addition to economy, there is to be considered such items as comfort, relative humidity, actual temperatures, mean radiant temperatures, surface temperatures, and the esthetic effect on personnel of both systems.[3] These comparative figures are indispensable to the design heating engineer, for every building is different, and the operating requirements are often different.

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[3] ______, Heating Ventilating Air Conditioning Guide 1947 (New York: American Society of Heating and Ventilating Engineers, 1947), 1279 pp.

II APPARATUS

The investigation consisted of three major compilations of data: (1) outside weather conditions, (2) conditions inside each room, and (3) steam room data on the transfer of heat from the steam to the water pumped to the rooms. This all had to be correlated by simultaneous readings. This was expedited by recording instruments. A twenty-four hour electric recorder was used to determine the outside temperatures, both wet and dry bulb. This was a Brown Instrument Company instrument, which was mounted near the building, but out in the weather.

For the inside readings, a sling psychrometer was used to ascertain the wet and dry bulb temperatures, and a Globe Thermometer was used to determine the Mean Radiant Temperature. This Vernon thermometer is made of a thin copper sphere six inches in diameter, coated with black paint and lamp black to approach unity for an emissivity factor.[4] A standard glass mercury thermometer is inserted into this globe, so that the mercury bulb is at the center of this sphere. Surface temperatures of the walls, floors, and ceilings, including windows, were determined by an instant reading surface pyrometer. The pyrometer used was an Alnor

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[4]C. G. Warner and T. Bedford, The Globe Thermometer in Studies of Heating and Ventilation (Industrial Health Research Board, 1934), pp. 458-473.

Pyrocon, type 4000, as manufactured by the Illinois Testing Laboratories in Chicago, with a type 4040 thermocouple for temperatures under 800 °F. A light aluminum stand was used to support thermometers at different levels from the floor and also to support the globe thermometer, which was in all cases placed in the approximate center of the room. As these Vernon thermometers required thirty minutes to stabilize, several stands were used to facilitate simultaneous readings in different rooms.

In the convertor room, recording instruments made simultaneous readings possible. Two condensate meters were located in the basement convertor room to determine the steam required by both systems individually. Pressure gages in the steam lines were sufficient to determine the steam pressure in the convertor, and thermometers in both the hot water and steam lines were used in both piping layouts for the differential heating systems. On the steam main was attached a Barrel calorimeter used to ascertain the quality of the steam supplied to the convertors.

The readings of these instruments are sufficient to determine the heat supplied to the building and the approximate heat dispensed by the building as exposure losses.[5]

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[5]William H. McAdams, Heat Transmission (New York and London: McGraw-Hill Book Company, Inc., First Edition, 1933) 383 pp.

III PROCEDURE

To eliminate the effect of solar radiation or to reduce it to a negligible figure, the inside readings for a comparison of room conditions were taken on very cloudy days, during, before, or just after a rain. The coldest day on which data was compiled was January 31, 1948, when the minimum temperature was 25 °F and the maximum 30 °F, giving $37\frac{1}{2}$ degree days, considering that no heat would be required if the outside temperature average were 65 °F.

The outside temperatures recorded on circular charts were taken by the recording instrument continuously, so that inside data at any time could be referred to outside conditions at that particular instant. Thus, all inside readings had to be timed so that the reference to the recording charts might be effected.

Representative rooms rather than apartments were chosen for study. Thus, rooms with similar exposure and size were used in both sides of the building (panel and convection).

The first readings taken in the convertor room were all compiled and timed, and then room data was compiled. These readings in the rooms were scheduled as nearly as practical to give simultaneous results. On the first floor, a stand was placed in one room in both sides of the building, so that there was a stand in both a panel heated room and a convection heated room at the same time. While the Vernon thermometers were given sufficient time to stabilize, surface temperatures of all the walls, ceilings, and floors in these rooms were taken along with the wet bulb temperature readings. An average temperature for each surface was tabulated by bringing the pyrometer tip in contact with that particular surface at several uniformly spaced points. This gave a good average surface temperature, especially around windows where a variance of a few inches changed the surface conditions.

The aluminum stands to which the thermometers were attached were placed in all rooms in the approximate center of the space, that is, equidistant from opposite walls. Permanent arms were fixed to the upright member, so that all thermometers would remain at the same height in every room. Three calibrated mercury thermometers were used: one six inches from the floor, another four feet from the floor, and the third seven and a half feet from the floor. The distance was measured from the floor to the mercury bulb of each thermometer. This presented a good picture of the temperature gradient in the room tested. The Vernon thermometer, also hung from this stand, was suspended so that its center was located two and one half feet from the floor, the approximate locus of a seated person.[6] By use of the Vernon thermometer, the radiant-convective temperature, as it is called, was

[6] _______, Heating Ventilating Air Conditioning Guide 1947 (New York: American Society of Heating and Ventilating Engineers, 1947) 1279 pp.

ascertained.[7] This was used to determine the radiant temperature, as nearly as practicable, as that sensed by a seated human body.

At this same level, a shielded thermometer was hung to the stand. The shield consisted of a ventilated aluminum foil cubicle, designed to eliminate the radiant effects and to record the true air temperature. However, it consistently produced the same reading as an ordinary thermometer in the same position.

After one room was completed, the stands and instruments were moved to another room on the same floor and the same readings as before were repeated. When the first floor was finished, the instruments were all moved to the eighth floor, and readings corresponding to those of the first floor were recorded. Then the fifth floor was the object of the investigation, so that a mean as well as the extremes might be represented by the survey.

The fifth floor was used to present a representative of the other floors, and the first floor was selected to show the effect of the non-heated basement on room conditions, and the eighth floor readings to show the effect of the exposed horizontal roof on the heating requirements. On all floors, excepting the eighth, in the panel heated division, there are panels both above and below in the ceiling and floor. How-

[7] _______, <u>Heating Ventilating Air Conditioning Guide 1947</u> (New York: American Society of Heating and Ventilating Engineers, 1947), 1279 pp.

ever, for the eighth story, there is no ceiling panel. Thus, the heating load of the panel in the floor of the eighth story is much larger than that of the other floors. After all the room data had been compiled, the convertor room readings were all repeated so as to give an average of this data and the steam requirements during this period required to obtain the inside room data.

This data was only taken when the weather conditions were favorable, that is, when there was no sun and a heavy overcast.

However, daily, for a twenty day period during an average of the heating season, all convertor room data was taken so that the twenty-four hour steam requirements of each heating system might be compared under various days of different temperatures and humidities. This twenty day period extended from January 31, 1948, to February 19, 1948, giving an overall degree day average of 22.4 degree days per day. During this period, the highest temperature recorded was 66 °F on February 18, 1948, whereas the lowest was set on January 31, 1948, at 25 °F.

IV DISCUSSION

The large number of variables involved in this type of an investigation makes an accurate comparison of results difficult. This survey was conducted while the building was actually in use and being occupied. It is a well-known fact that people like certain rooms warmer than others, and that different people desire that the same room be held at a different temperature for their own personal comfort. No control over the rooms was attempted, but it was assumed that each room was maintained at a temperature that any other occupant would have desired or very nearly that desired by another. That is to say, that if the occupant of a panel heated apartment had moved into a corresponding convection heated apartment (or conversely), no change in the heating would be made. Establishing this premise, it is feasible to compare the two systems by the prevailing conditions in the different sections.

Although the highest dry bulb temperature recorded was in a panel heated apartment, generally the air temperature requirements for the panel heated apartments were lower than those of the corresponding convection heated rooms (see Figure 1). The average temperature of the panel heated rooms was one degree less than that of the compared system. A greater distinction is shown in the temperature gradient charts (see Figures 2 and 3). The most desirable gradient on these graphs would be represented by a vertical line showing no variation in the dry bulb temperature from the floor to the ceiling. Since no absolute physical law affects this gradient, the points of this plot were connected by straight lines. Here again the greatest deviation from the desirable was in a panel heated room, this being on the fifth floor in a room where the floor panel was completely shut off, deriving its only heat from the overhead panel where the temperature was greatest in that particular room. The temperature gradients of the panel heated rooms are much more desirable than those of the compared units, the air near the floor being closer to the average room temperature. In the panel heated rooms, the air near the floor was warmer on the average than that of corresponding convection rooms. This is desirable, as one occupying a room is either standing or seated (more often seated) and occupying more effectively the lower half of the room.[8] Thus, in a panel heated apartment, one is less apt to feel a cool sensation around his feet.

The mean radiant temperature as indicated by the overall area average (see calculations) was higher in the radiant section by approximately two (2) degrees Fahrenheit. However, the MRT as indicated by the Vernon thermometer (see calculations) was approximately the same in all apartments. Thus, the actual surface temperatures in the panel heated rooms

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[8] ________, Heating Ventilating Air Conditioning Guide 1947 (New York: American Society of Heating and Ventilating Engineers, 1947), 1279 pp.

were recorded higher than in the convection rooms. In fact, in one room on the fifth floor, the kitchen of apartment 57 (see sample data sheet) had a floor temperature of 102 °F. This, being too hot for foot contact and occurring in one room only of one apartment, was considered abnormal.[9]

Neither system offers absolute control of the moisture content in the air. It is very desirable to control this.[10] This is regulated only by the number of windows left open and the amount of fresh air entering the room. The relative humidity, although in the comfort zone,[11] was of such wide variance that no average would be representative, but generally the same approximate relative humidity was maintained by both units. That is to say, the necessary moisture for comfort was maintained by both systems and in no case was the air too dry for comfort (see Figure 4).

Of all the data assembled, the most concrete results were evidenced by the overall economy of the panel heating system, as contrasted with that of the convective layout. Although the degree day is not an absolute indication of the

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[9] _______, Heating Ventilating Air Conditioning Guide 1947 (New York: American Society of Heating and Ventilating Engineers, 1947), 1279 pp.

[10] B. F. Raber and F. W. Hutchinson, op. cit.

[11] _______, Heating Ventilating Air Conditioning Guide 1947 (New York: The American Society of Heating and Ventilating Engineers, 1947), 1279 pp.

heating requirements, it is very definitely a good indication of how cold one day is relative to another. The degree day is computed by taking the mean of the maximum and minimum temperatures for a twenty-four hour period and subtracting this mean from sixty-five, the temperature at which it is assumed that no heating will be required. Using the degree day for a comparison of the outside weather conditions (see Figure 5), the steam requirements can be compared on a basis fairly representative of the heating load. During the twenty day period of the test for every day, excepting one, the steam requirements of the panel system were consistently lower than the convection system. On that one day, the steam requirements were identical for both units.

The average steam required per day was 6,945 pounds for the radiant and 10,035 pounds for the convective apartments. The average degree days per day during this period was 22.4 (see tables). This is an average daily saving of 30.8% by the radiant system as contrasted to its competitor, a difference of 3,090 pounds of steam per day (see Figure 5). That is, the steam requirement of the panel heated system was only 69.2% of the convective. If these figures were reduced to the same floor space by a linear variation, the savings as evidenced by the panel heated unit would be greater than the figures stated above. With this correction, the average steam requirements of the panel heated space reduced to the same floor space would be 6,230 pounds per day for the radiant system, showing a saving of 3,805 pounds of steam per day or a 37.9% saving over the convective needs. The steam was supplied to the convertors at an average temperature of 225 °F and a quality of 98% during this test period. The average heat delivered to the two sections was for a daily average 7,121,250 BTU for the radiant and 9,648,200 BTU for the other (see tables and Figure 6). This makes the panel heated apartments 30.8% to 37.9% more efficient to operate in winter than the convection heated apartments.

V SUMMARY

From an overall survey of all the data assembled, it seems that to say that radiant heat is purely radiant is a misnomer. In fact, it may be illogical to say that the majority of the heat sensed by one in a radiant room is radiant. The temperature gradients in the radiant apartments were more desirable than those of the convective units, but was this due to radiation or to a more efficient convective circuit of very large area, both in the floor and in the ceiling, making the air temperatures near these surfaces higher? No effective study of convective currents was made. A velometer was used which indicated no air movement beyond 16 feet per minute, equal to still air for all practical heating purposes. Take, for example, a typical room in both divisions, which has only one outside wall, with windows in that wall. The cooling of the air occurs around this window, and the air settles to the floor until heated by whatever system is installed. In the case of the convection heated room, the air is heated just below the window by the convection unit, which reverses the air flow, lifting the warm air to the ceiling, causing it to move down the far wall after it has been cooled. This is most definitely convection heating. This differs markedly from the panel heating. The air is cooled along the outside wall and drops to the floor. It passes over a large area where it is heated in a large mass, causing it to rise to the ceiling in mass area, where it is further heated by the ceiling panel, this time building up a bank of warm air that gradually expands to the center of the room giving this panel heated room better temperature gradient results. If this is true, then a large part of the heat delivered to the room is convective rather than radiant, even in the supposedly radiant heated apartments. It appears logical to assume that these panels in the floor and ceiling are very large convective units, as well as large radiating panels, having for an exposed surface the entire ceiling or floor. With such a large area in contrast to the smaller area of one convective unit, the heat transfer could only be more efficient in the radiant division, neglecting the probable increase in radiant effects.

The name, "Panel Heat", seems more applicable and certainly less a misnomer.

VI CONCLUSIONS

1. Once a panel heating system is installed, it is more efficient to operate than a convective system.

2. Neither system has a control of the moisture, but the relative humidity of the two systems during the investigation was within the comfort zone.

3. Panel heated rooms operate at a dry bulb temperature less than the convective rooms.

4. The temperature gradients (air stratification) in panel heated rooms are more desirable than those of the convection heated rooms.

5. The mean radiant temperature of the panel heated rooms is approximately 2 °F higher than those of the convective division in average operating conditions.

6. Neither system is purely radiant or convective, as the name implies. The convection heat system is more effectively convective than the panel heated circuit is radiant.

7. Since there are no physical heating units installed within the confines of the panel heated rooms, they have more

esthetic appeal than the other tested system. Floor space is taken up by the convective heater. Unless the convection heated rooms are kept spotlessly clean and air filtered, streaks on the walls will appear around the heater, where concentrated convective currents rise from the heater.

8. The major disadvantage of the panel heated section is the lack of complete independent control of the room temperature due to the overhead panels which heat the room above and below, but are only controlled by valves on the floor above. This was particularly true on the odd numbered floors, as the eighth floor used a maximum of heat due to the unheated roof, which in turn was more than enough to heat the rooms on the seventh floor. Thus, the occupants on the seventh floor had their heat completely off even on the coldest day, requiring the sixth floor occupant to maintain his system open to the maximum. The control alternated successively downward.

9. The panel heat system has a particular disadvantage in that there is a marked thermal lag in the control of the system, for time is consumed in bringing the 5-inch slab of concrete to the operating temperature from cold conditions. Similarly, to reduce the heat delivery to the rooms, there is a lag, for the mass of concrete must

cool after the hot water flowing through the pipes has been discontinued. This disadvantage is most in evidence during the fall and spring months, when the heating requirements are less apt to be stable and control is more desirable. This lag is of particular disadvantage when separate room control is desired for night occupancy. That is, cooling a panel heated room without opening windows requires a considerable amount of time. This is a general consensus of opinion, as expressed by the occupants of the building.

BIBLIOGRAPHY

Adlam, T. Napier, Radiant Heating. New York, N.Y.: The Industrial Press, 1947. 472 pp.

Brabbee, C. W., Heating and Ventilating, A Handbook for Engineers. New York: McGraw-Hill Book Company, Inc., First Edition, 1927. 332 pp.

Dye, Frederick W., Heating and Hot Water Work. London: E. & F. N. Spon, Ltd., New York: Spon and Chamberlain, 1932. 192 pp.

Jakob, Max and Hawkins, G. A., Elements of Heat Transfer and Insulation. New York: John Wiley and Sons, London: Chapman and Hall, Ltd., Fourth Printing, 1947. 169 pp.

Kratz, Alonzo P. and Harris, Warren S., A Study of Radiant Baseboard Heating in the I B R Research Home. Urbana, Illinois: University of Illinois Engineering Experiment Station Bulletin No. 358, 1945. 33 pp.

Laurens, Henry, The Physiological Effects of Radiant Energy. New York: The Chemical Catalog Company, Inc., 1933. 610 pp.

Mackey, C. O., Radiant Heating and Cooling. Ithaca, N.Y.: Cornell University Engineering Experiment Station Bulletin No. 32, 1943. 29 pp.

McAdams, William H., Heat Transmission. New York and London: McGraw-Hill Book Company, Inc., First Edition, 1933. 383 pp.

Raber, B. F. and Hutchinson, F. W., Panel Heating and Cooling Analysis. New York: Johns Wiley and Sons, Inc., London: Chapman and Hall, Ltd., 1947. 208 pp.

Rubert, Kennedy F., Heat Emission from Radiators. Ithaca, N. Y.: Cornell University Engineering Experiment Station Bulletin No. 24, 1937. 53 pp.

_______, Heating Ventilating Air Conditioning Guide 1947. New York: American Society of Heating and Ventilating Engineers, 1947. 1279 pp.

Warner, C. G. and Bedford, T., The Globe Thermometer in Studies of Heating and Ventilation. Indistrial Health Research Board, 1934. pp. 458-473.

APPENDIX I

SAMPLE CALCULATIONS

1. Mean Radiant Temperature, MRT_V, as indicated by the Vernon thermometer

$$\left(\frac{T_s}{100}\right)^4 = 4(t_g - t_a) + \left(\frac{T_g}{100}\right)^4$$

where: T_s is the value to be calculated, degrees Rankin

t_g is the temperature recorded by the globe, degrees Fahrenheit

t_a is the ambient temperature, degrees Fahrenheit

T_g is the globe temperature in degrees Rankin

Apartment 57 Living Room

$t_a = 80\ ^\circ F$

$t_g = 78.5\ ^\circ F$

$T_g = 78.5 + 460 = 538.5\ ^\circ R$

$$\left(\frac{T_s}{100}\right)^4 = 4(80 - 78.5) + (538.5/100)^4$$

$$T_s = 539\ ^\circ R = MRT_V = 79\ ^\circ F$$

2. Mean Radiant Temperature, MRT_A, as indicated by actual surface readings

Apartment 84 Living Room

Surface	Temp. °F	Area Sq. Ft.	Product
North	71	224	15,900
East	64	113.5	7,260
South	72	224	16,120
West	68	91	6,190
Ceiling	68	364	24,800
Floor	73	364	26,400
Glass-west	57	45	2,562
Glass-east	57	22.5	1,280
Total		1448.0	100,512

MRT_A = Product total divided by area total

$= 100,512/1448.0 = 69.5\ °F$

3. Heat Supplied to the Convertor

$$Q = W(h_g - h_f)$$

where: Q is the BTU supplied

W is the weight of steam per day in pounds

h_g is the enthalpy of the steam entering the convertor

h_f is the enthalpy of the condensate leaving the convertor

Convective Requirements for January 31, 1948

$$Q = 13,000\ (1145 - 174)$$

$$= 12,620,000\ \text{BTU/24 hrs.}$$

4. Estimate of heat load for January 31, 1948, degree day of 22.4

Heat Dispensed

	Outside Surface of Builsing Sq. Ft.	Temperature Differential F	Conductivity	Product BTU/hr.
Ceiling	3,038	40	.32	33,800
Floor	2,948	25	.39	28,700
Walls	34,948	40	.32	447,000
Glass	8,172	40	1.13	369,000
Infiltration	.018(219,000)40			157,000

Total BTU/hr. 1,035,500

Estimated Load BTU/24hrs 24,852,000

Heat Supplied BTU/24 hrs.:

Radiant	11,620,000
Convection	12,620,000
Total	24,240,000

SAMPLE DATA SHEET

BUILDING SOUTH HEATING UNIT RADIANT DATE JAN. 31, 1948

TIME	OUTSIDE READINGS				INSIDE READINGS																			
Hr. &. Min.	Dry Bulb °F.	Wet Bulb °F.	R.H. %	Weather Description	Floor No.	Apt. No.	Room	Dry Bulb °F.	Wet Bulb °F.	R.H. %	Gradient Dry Bulb Ft. From Floor			SURFACES										Air Movement
											0.5	4.0	7.5	Walls North	East	South	West	O.G.	Ceiling	Floor	M.R.T.	Vernon °F.	Shield °F.	
P.M.																								
2:45	29.5	26	62	CLOUDY	5	57	L	79	64	44	72.5	80	81.25	76	75	65	86	57	82	72	24	78	79	X
2:45	29.5	26	62	CLOUDY	5	57	K	80	64	42	77	76	78	90	80	86	70	60	89	82	83	81	80	X
3:20	29.0	26.5	75	CLOUDY	8	85	L	76	61.5	43	77	75	78	74	73	70	66	55	65	93	74	75	77	X
3:20	29.0	26.5	75	CLOUDY	8	85	B/W	76	59.0	35	71	72	70	70	61	70	65	57	69	90	70	76	76	X

SAMPLE DATA SHEET

NORTH AVENUE APT. HOUSE
CONVERTOR ROOM DATA

TIME *AFTERNOON* DATE *JAN. 31, 1948*

	TIME IN HRS. AND MINS.	CONDENSATE METER READINGS	HOT WATER TO THE ROOMS	RETURN WATER FROM ROOMS (BY-PASS)	HOT WATER FROM THE HEAT EXCHANGER	SUPPLY STEAM TEMPERATURE	RETURN STEAM TEMPERATURE	BY-PASS VALVE SETTING	INLET STREAM PRESS	THROTTLED STEAM PRESSURE	STEAM VALVE SETTING	UNIT
UNITS		#	F	F	F	F	F	NO.	PSI	PSI	NO.	TYPE
	12:30	725,506	181	173	189	222	206	0	40	15	1/2	CONVECTION
	12:30	853,400	133	113	176	222	216	3/4	40	15	c	RADIANT
	5:25	728,200	182	173	188	221	206	0	40	14	1/2	CONVECTION
	5:25	855,800	133	113	175	221	216	3/4	40	14	c	RADIANT

APPENDIX II

TABLE I: Steam Consumption Per Day

Date		Outside Temperatures Max. °F	Min. °F	24Hr. Ave.	Degree Day	Radiant Pounds Steam Per Day	Convec. Pounds Steam Per Day	Radiant Corrected*
Jan. 31	Dry	30.0	25.0	27.5	37.5	12,100	13,000	10,850
	Wet	27.5	22.0	24.75				
Feb. 1	Dry	38.5	25.0	31.75	33.25	10,900	13,000	9,830
	Wet	30.5	22.5	26.5				
Feb. 2	Dry	42.0	27.0	34.5	30.5	10,500	12,700	9,420
	Wet	32.5	23.5	28.0				
Feb. 3	Dry	51.5	36.5	44.0	21.0	9,000	11,800	8,070
	Wet	43.0	32.0	37.5				
Feb. 4	Dry	52.0	44.0	48.0	17.0	8,800	10,700	7,890
	Wet	47.5	41.0	44.25				
Feb. 5	Dry	48.5	40.0	44.25	20.25	5,700	12,900	5,120
	Wet	47.0	37.5	42.25				
Feb. 6	Dry	48.0	39.0	43.5	21.5	6,500	6,500	5,830
	Wet	44.0	37.5	40.75				
Feb. 7	Dry	45.0	38.5	41.75	23.25	9,000	10,500	8,070
	Wet	43.5	37.0	40.25				
Feb. 8	Dry	79.0	36.5	57.75	7.25	7,400	10,400	6,640
	Wet	77.0	38.5	57.75				
Feb. 9	Dry	32.5	28.0	30.25	34.25	10,400	11,700	9,330
	Wet	30.5	26.5	28.50				
Feb. 10	Dry	35.0	28.0	31.5	33.50	9,200	11,100	8,250
	Wet	32.5	26.0	29.25				
Feb. 11	Dry	39.0	30.0	34.50	30.50	11,000	12,500	9,860
	Wet	37.0	26.5	31.75				
Feb. 12	Dry	52.5	39.5	46.00	19.00	8,200	11,500	7,350
	Wet	51.5	37.5	44.5				

*Correction made as noted in DISCUSSION.

Date		Outside Temperatures Max. °F	Min. °F	24Hr. Ave.	Degree Day	Radiant Pounds Steam Per Day	Convec. Pounds Steam Per Day	Radiant Corrected
Feb. 13	Dry	62.0	33.0	47.5	17.50	5,400	8,000	4,840
	Wet	57.5	29.0	43.25				
Feb. 14	Dry	41.0	28.0	34.5	30.50	7,900	10,700	7,080
	Wet	34.0	24.5	29.25				
Feb. 15	Dry	51.0	40.0	45.50	19.50	2,700	10,600	2,420
	Wet	41.0	33.0	37.00				
Feb. 16	Dry	56.0	41.0	48.5	16.50	7,500	9,300	6,720
	Wet	49.0	38.0	43.5				
Feb. 17	Dry	61.0	39.0	50.0	15.0	2,800	5,900	2,510
	Wet	50.0	36.0	43.0				
Feb. 18	Dry	66.0	49.0	57.5	10.5	3,000	3,500	2,690
	Wet	52.5	44.0	48.75				
Feb. 19	Dry	64.0	47.0	55.5	9.5	900	5,400	807
	Wet	52.0	44.0	48.0				
Average					22.4	6,945	10,035	6,230

TABLE II: Steam Heat Delivered Per Day.
B.T.U. per Day.

Degree Day	Radiant	Radiant Corrected*	Convection System	
37.5	11,620,000	10,420,000	12,620,000	
33.25	10,500,000	9,460,000	12,610,000	
30.5	10,050,000	9,000,000	12,300,000	
21.0	8,490,000	7,600,000	11,450,000	
17.0	8,380,000	7,500,000	10,400,000	
20.25	5,430,000	4,870,000	12,590,000	
21.5	6,310,000	5,660,000	6,340,000	
23.25	8,560,000	7,680,000	10,250,000	
7.25	7,060,000	6,340,000	10,190,000	
34.25	9,950,000	8,930,000	11,390,000	
33.5	8,800,000	7,880,000	11,200,000	
30.5	10,550,000	9,450,000	12,110,000	
19.0	7,980,000	7,160,000	12,130,000	
17.5	5,130,000	4,600,000	7,770,000	
30.5	7,550,000	6,770,000	10,400,000	
19.5	2,579,000	2,310,000	10,320,000	
16.5	7,130,000	6,380,000	9,050,000	
15.5	2,662,000	2,390,000	5,740,000	
10.5	2,840,000	2,550,000	3,339,000	
9.5	854,000	518,000	765,000	
22.4	7,121,250	6,373,400	9,648,200	Average

*Correction made as noted in the DISCUSSION.

TABLE III: Cost of Operation.
Steam supplied at an average temperature of 225 °F and a quality of 98%. Steam delivered to the building at a flat rate of $.85 per thousand pounds.

Degree Day	Cost per Day Convective	Radiant	
37.5	$ 10.30	$ 10.30	
33.25	11.05	9.26	
30.5	10.80	8.93	
21.0	10.00	7.65	
17.0	9.10	7.48	
20.25	10.95	4.84	
21.5	5.53	5.53	
23.25	8.93	7.65	
7.25	8.84	6.29	
34.25	9.94	8.84	
33.5	9.44	7.83	
30.5	10.62	9.35	
19.0	9.79	6.96	
17.5	6.80	4.59	
30.5	9.10	6.72	
19.5	9.02	2.29	
16.5	7.90	6.37	
15.5	5.02	2.38	
10.5	2.98	2.48	
9.5	4.59	.77	
22.4	$ 8.58	$ 6.33	Average per Day

TABLE IV: Tabulation of Data for a Sample Day, January 31, 1948.

Dry Bulb Readings
Degrees Fahrenheit

Outside	Panel	Convection
29.0		74.0
28.5		76.0
28.5		75.5
28.5		72.0
30.0		77.75
30.0		75.0
30.0		77.0
30.0		70.5
29.5	80.0	
29.5	76.0	
29.0	75.0	
29.0	72.0	
29.0	73.0	
29.0	76.0	
28.5	71.0	

Relative Humidity
Per Centage

Outside	Panel	Convection
70.0		37.0
75.0		41.5
70.0		37.0
70.0		49.0
62.0	44.0	
62.0	42.0	
75.0	43.0	
75.0	35.0	
80.0	59.0	
81.0	45.0	
78.0	37.0	42.0
80.0		29.0
80.0		58.0
80.0		60.5

TABLE IV: Tabulation of Data for a Sample Day, January 31, 1948. -(Concluded)

Mean Radiant Temperatures
Degrees Fahrenheit

CONVECTION

Inside Dry Bulb	MRT_A	MRT_V	
77.75	72.3	76.5	
70.50	71.7	77.0	
74.00	82.0	78.0	
75.50	69.5	74.0	
	73.88	76.39	Average

PANEL

Inside Dry Bulb	MRT_A	MRT_V	
80.0	74.4	78.5	
76.0	82.9	81.0	
75.0	73.7	75.0	
76.0	74.5	76.5	
71.0	71.3	71.0	
	75.3	76.4	Average

APPENDIX III

FIGURES

Figure 1.

INSIDE TEMPERATURES versus OUTSIDE TEMPERATURES
January 31, 1948

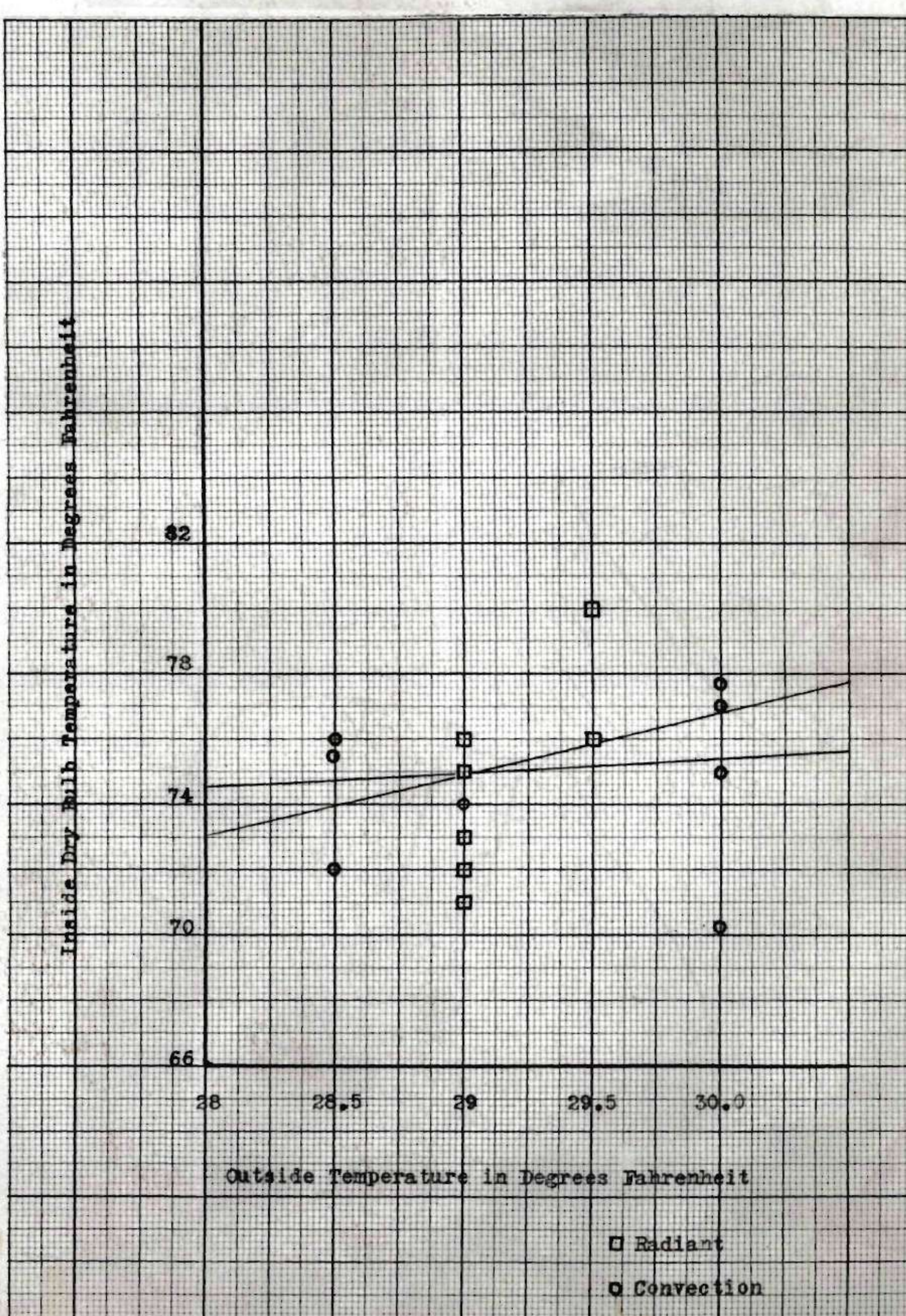

Figure 2.

TEMPERATURE GRADIANTS RADIANT APARTMENTS
January 31, 1948

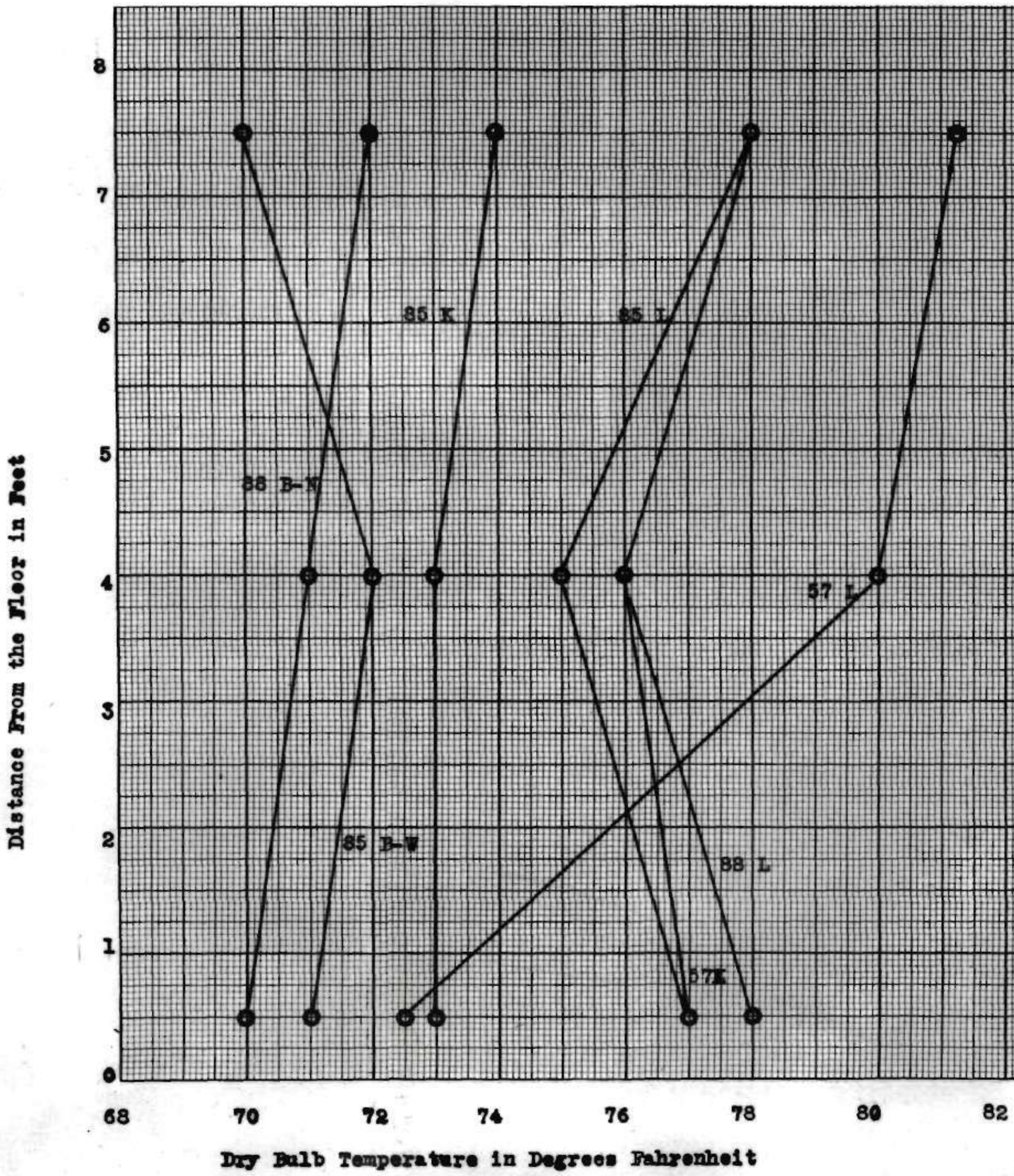

Figure 3.

TEMPERATURE GRADIENTS CONVECTION APARTMENTS
January 31, 1948

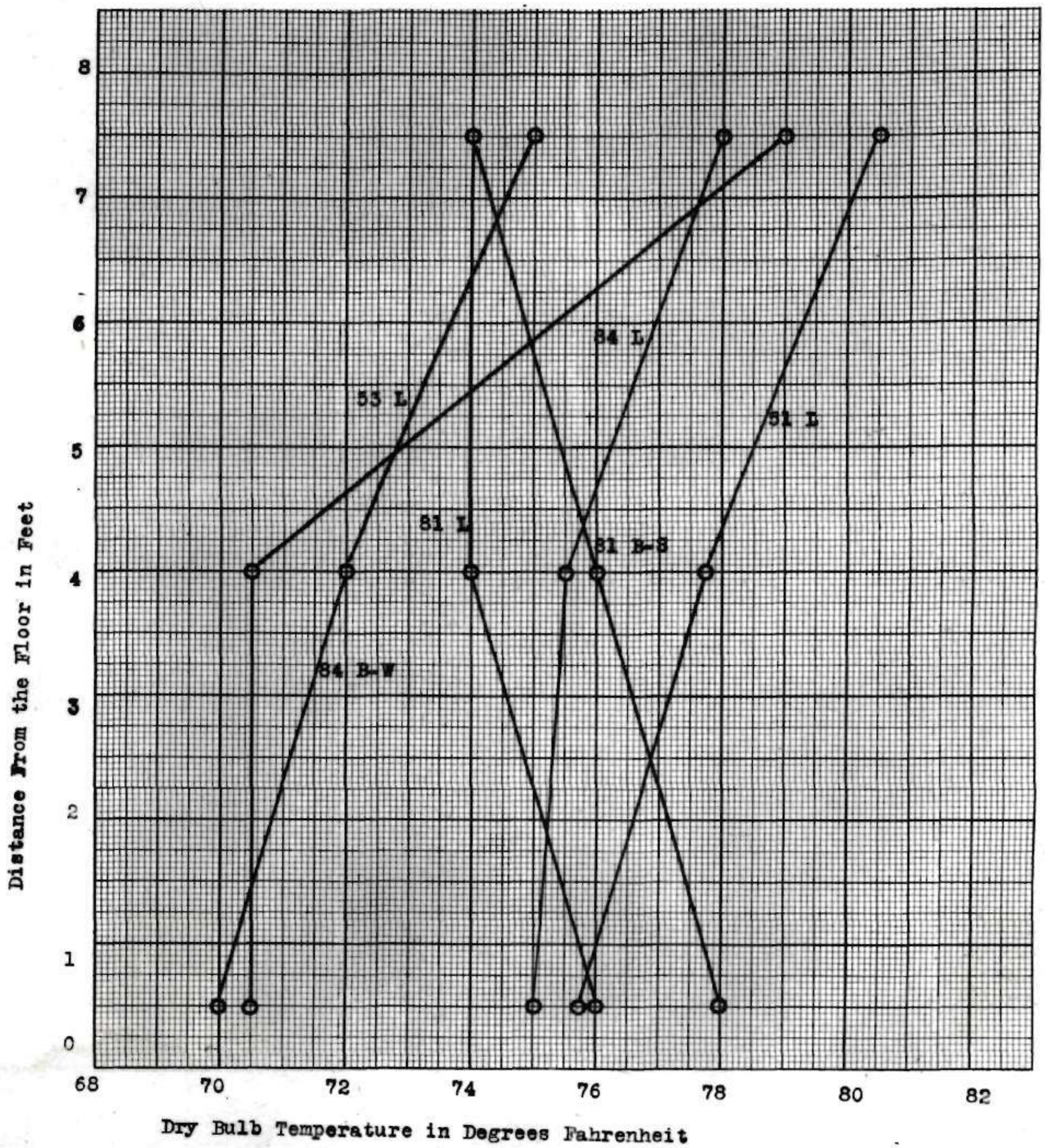

Figure 4.

RELATIVE HUMIDITY CHARACTERISTICS
January 31, 1948: 37.5 Degree Day, Max. Temp 30F,
Min. Temp. 25F

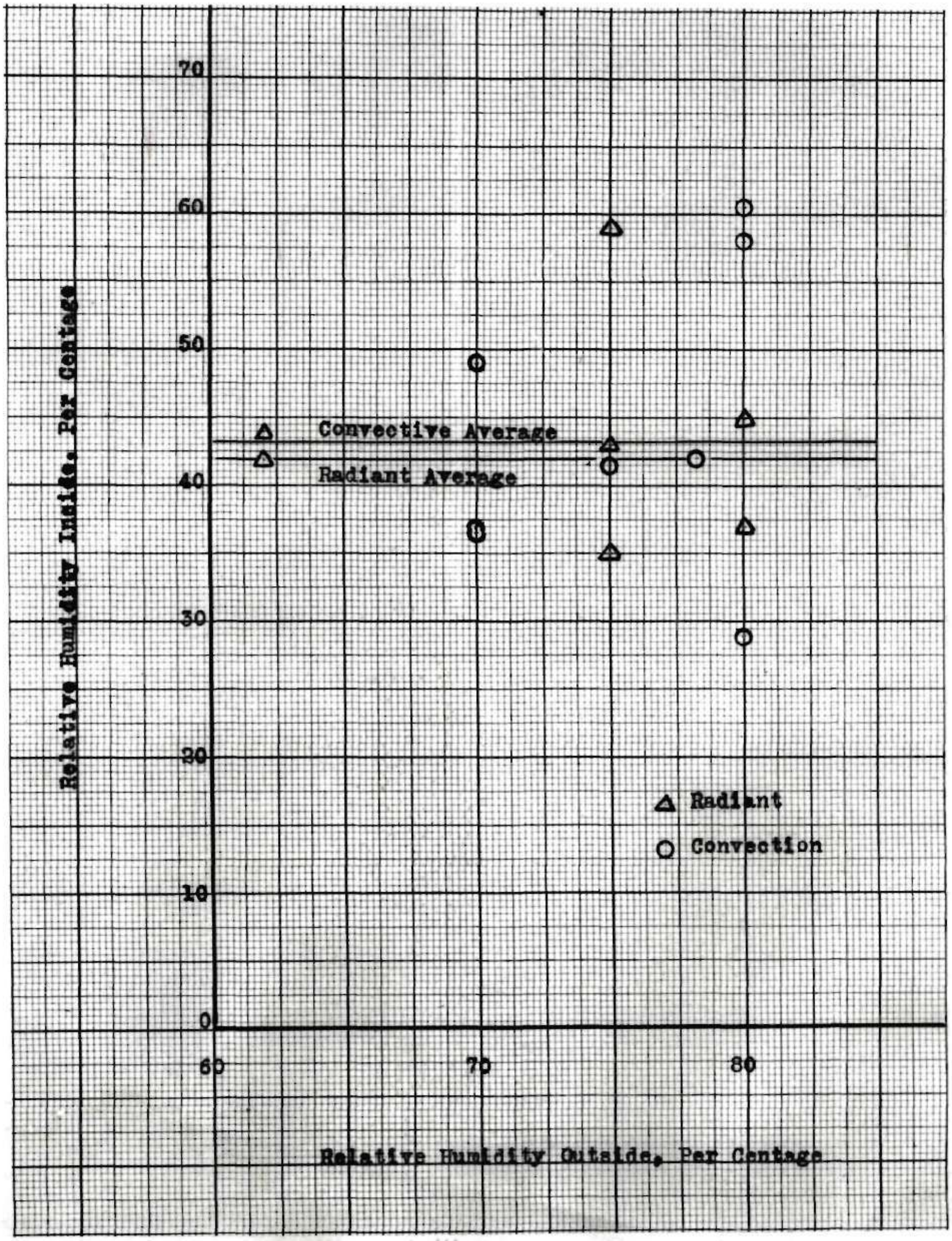

Figure 5.

STEAM CONSUMPTION CHARACTERISTICS
For 20 Day Period from Jan. 31 to Feb. 19, 1948

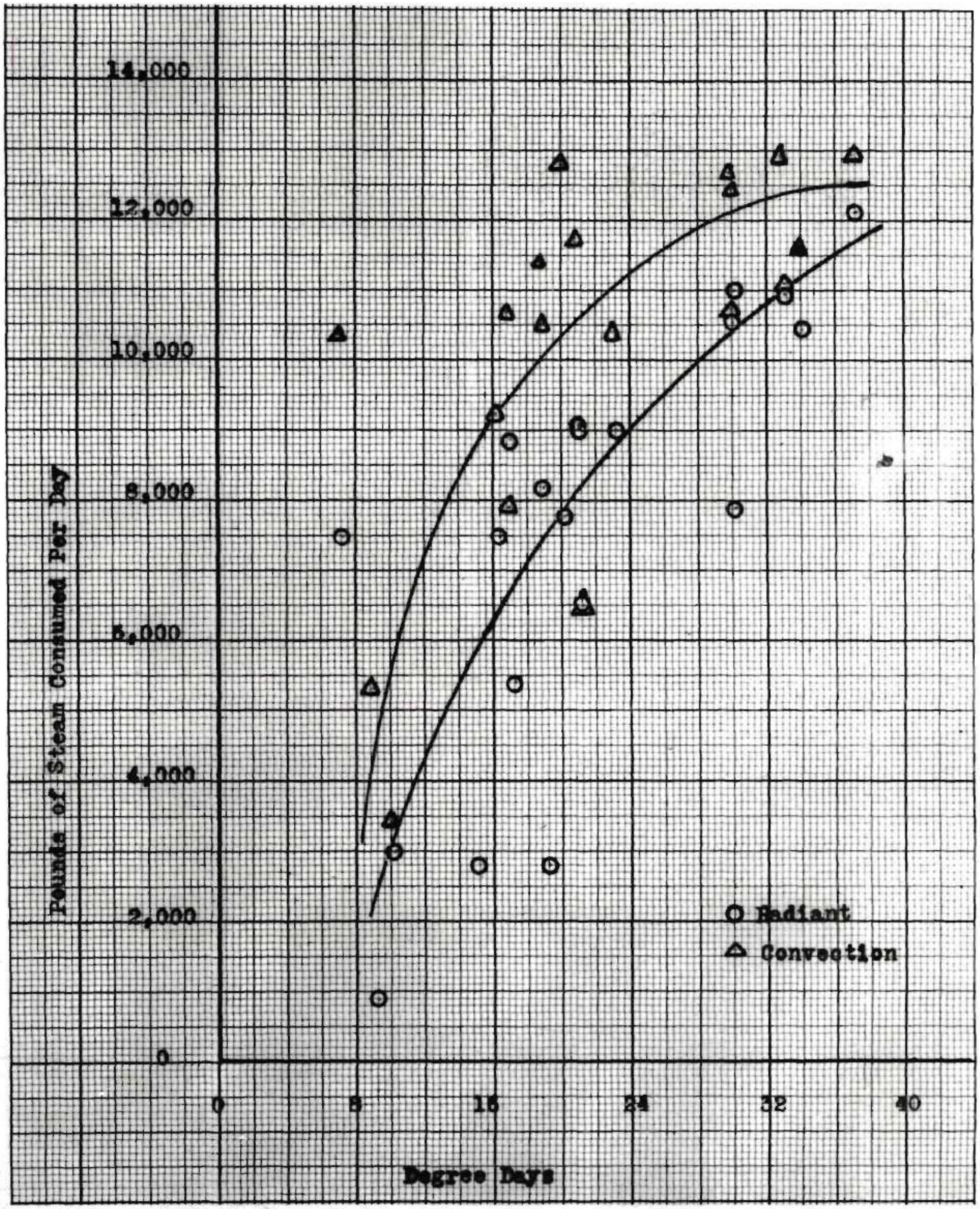

Figure 6.

HEAT CONSUMPTION CHARACTERISTICS
For 20 Day Period from Jan. 31 to Feb. 19, 1948

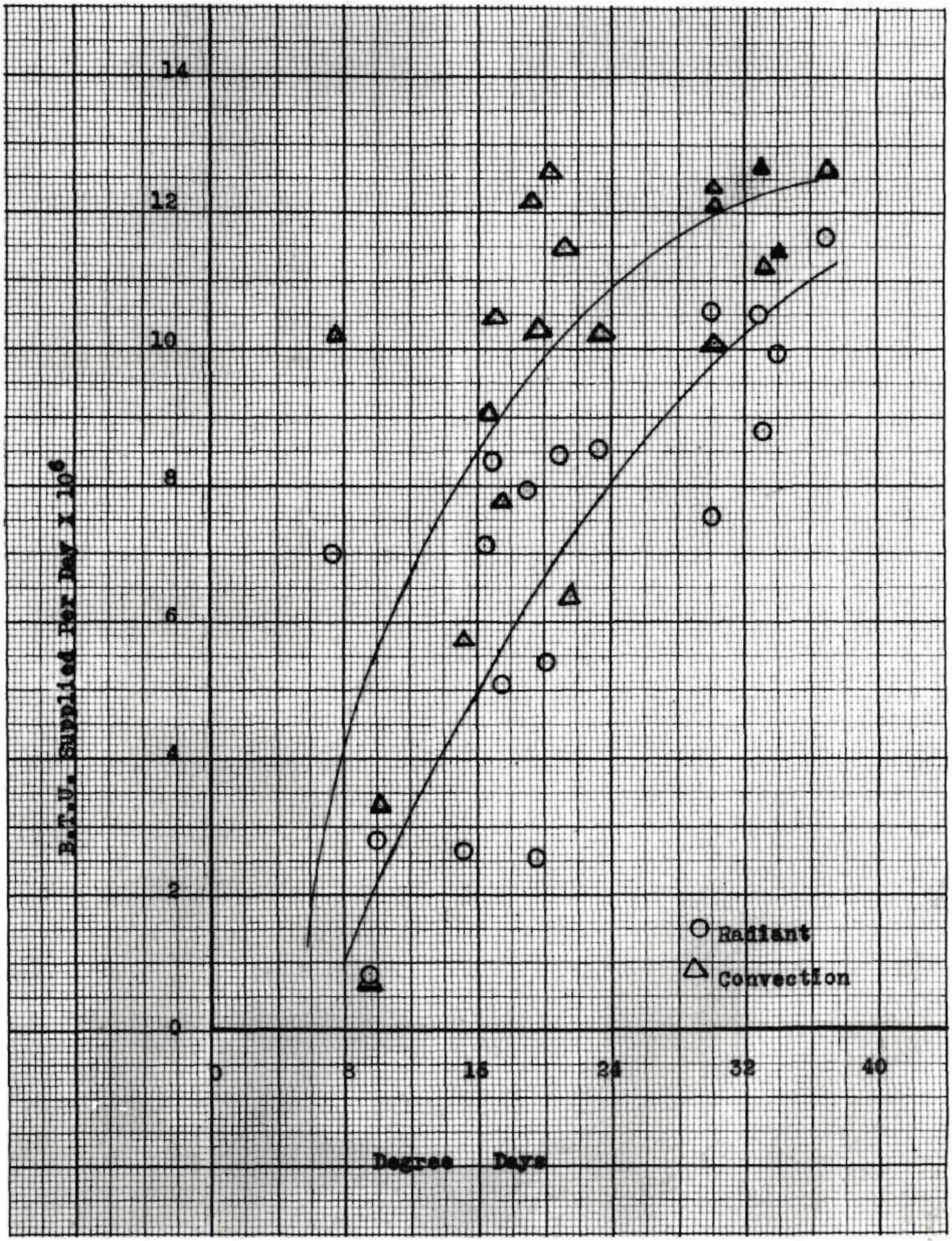

Figure 1. BURGE APARTMENT BUILDING

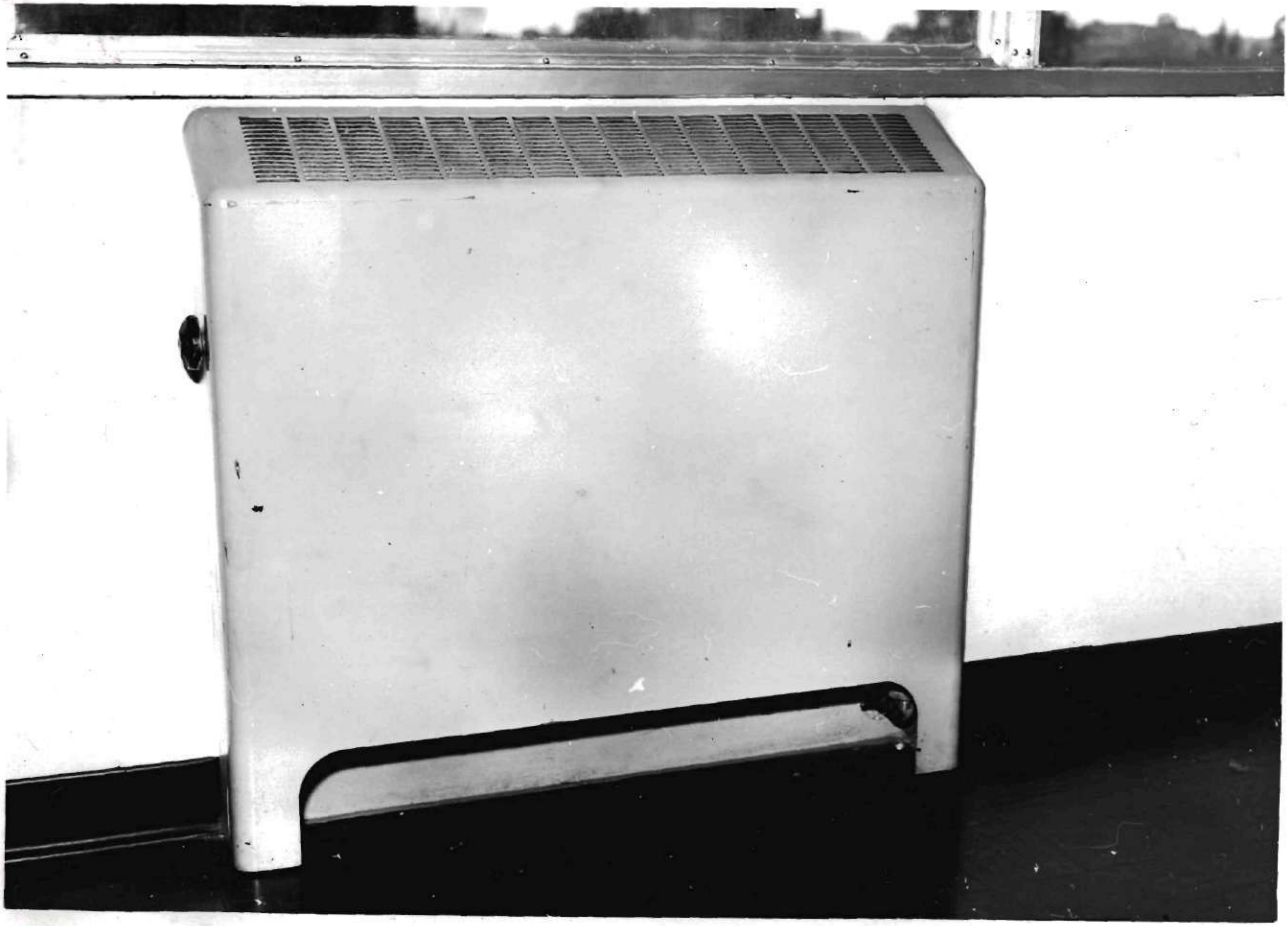

Figure 2. TYPICAL CONVECTION UNIT (INSTALLED)

Figure 3. TESTING APPARATUS

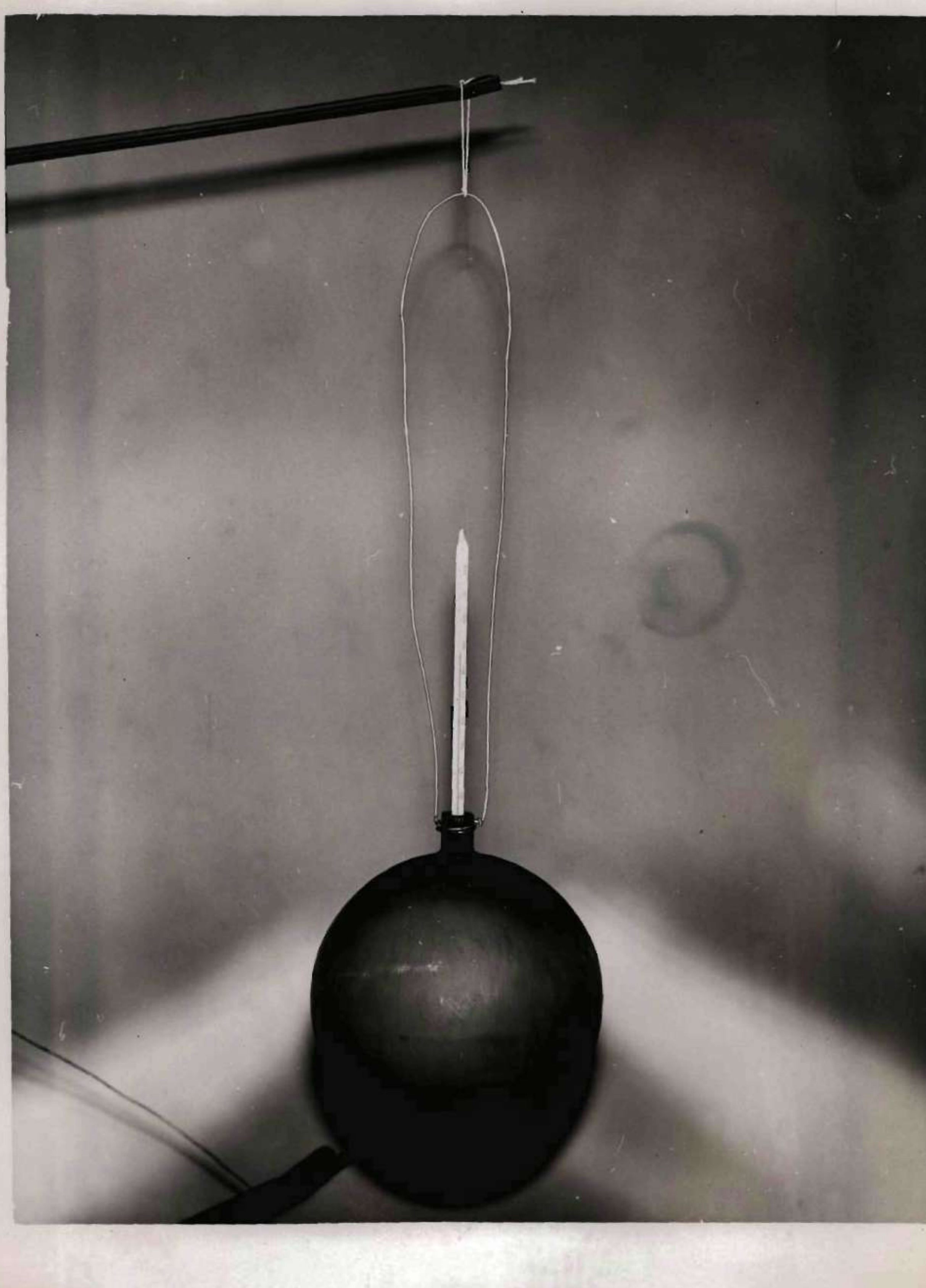

Figure 4. VERNON THERMOMETER